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A RESEARCH PROGRAM TO ESTABLISH DESIGN CRITERIA AND DEVELOP AN AEROSOL ULTRACENTRIFUGE

HEW-USPHS-NAPCA Contract No. 22-68-7

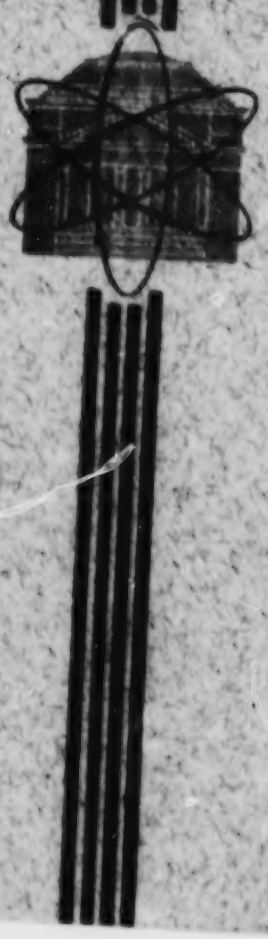
Interim Report

Period March 1, 1968 to June 30, 1969

Submitted to:

National Air Pollution Control Administration
Cincinnati, Ohio

By:
Fulvio A. Iachetta
Project Director



Research Laboratories for the Engineering Sciences

University of Virginia

Charlottesville

Report No. ME-3117-104-70U
January 1970

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RESEARCH LABORATORIES FOR THE ENGINEERING SCIENCES
SCHOOL OF ENGINEERING AND APPLIED SCIENCE
UNIVERSITY OF VIRGINIA
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INTRODUCTION

The work outlined briefly in this interim report, was conducted during the sixteen months period extending from the date of contract PH-22-68-7 award, March 1, 1968 through June 30, 1969 when the contract was renewed. Achievement of a mechanically sound machine occupied 75% of this period for reasons which will be clear from the following discussion. Serious flow pattern study was limited to the final four months covered by the report. In broad terms the time was expended in design analysis, detail component design and fabrication of an initial machine, design and construction of a suitable experiment including a suitable laboratory aerosol system, evaluation of mechanical performance and finally, experiments directed to delination of significant parameters influencing particle deposition pattern.

Project initiation was a preliminary design study into the specifics of an ultracentrifuge capable of classifying particles with sizes in the range 0.05-1.0μ with a throughput of 0.5 m^3/min while operating in the ambient. The physical size of such a machine had been estimated in the original proposal upon which this work was based. Further study of a much simplified and idealized flow model did not alter these dimensions though speed requirement was reduced to 40,000 rpm as a probable maximum. Design alternatives were studied with a nominal bowl inner diameter of 6.00 inches a machine length of approximately 30 inches. This design effort was begun with an intent to produce a purely laboratory device as an initial test vehicle, this choice having been made to take advantage of the existing high speed rotor test facility already available within the

from page 1

department of mechanical engineering and thereby reducing first costs and offering the prospect of earlier mechanical operation. Detailed design considerations, presented in greater detail in the following sections of this report, altered this approach and some prototype work was necessary in arriving at a first design. Special components requirement were slow of delivery and suitable materials difficult to obtain, events which combined to slow the first phase work.

Development of an experimental arrangement with sufficient flexibility to enable simultaneous study of internal flow pattern parameters and mechanical characteristics of the system was accomplished while awaiting complete machine assembly. Procedurely, experimental evaluation of the dynamical behavior and stability of the centrifuge was expected to be essentially a by-product of the work directed to particle deposition analysis. As is sometimes true with experimental research, events conspired to defeat this approach and concurrent evaluation of the mechanical and fluid mechanical characteristics was not possible. Serious dynamics and stability problems requiring of solution before any sampling work could be undertaken required the better part of a year for solution. While it is not possible to reproduce all the details of the work here, the sections which follow represent an attempt to provide at least a brief picture of the manner in which the project progressed.

DESIGN CONSIDERATIONS

General

Preliminary design study involved several factors peculiar to the ultimate machine configuration. A relatively simple machine had been envisioned but detailed analysis of such requirements as needed drive power, easy access to collected sample, sample collection provision, and sufficient flexibility to permit modification of inlet and outlet geometry altered the original concept considerably. Choice of a material was dictated ultimately by availability, machinability and cost. High maraging steels, titanium, fiber reinforced resins and aluminum 7075-T6 alloy were all considered. The aluminum alloy was chosen primarily due to availability, though of the materials considered it was the least strong. Actual dimensions for the experimental centrifuge bowl diameter was fixed by the size of extruded Al7075-T6 billet which could be purchased from a supplier. Final dimensions were, 6.500" inner diameter, 0.080" wall and a length of 30 inches. These dimensions permitted a maximum safe speed of 44,000 rpm at which speed stress in the centrifuge wall was estimated to be 60,000 psi which is 75% of the 80,000 yield strength claimed for the alloy.

With the material fixed, specific design of end closures, support bearings, drive and related matters was undertaken. Significant information relative to this detailed design is developed in the following sections.

Drive Power Requirements:

Drive power was determined to be overwhelmingly that needed to overcome viscous drag. The drag on the bowl is given by

$$\text{Drag, } D, = C_D \rho \omega^2 a^3 L \pi / g \quad \text{lbs.} \qquad (1)$$

and the power by

$$\text{Power, p, } = \frac{Dan}{63025} \text{ h.p.} \qquad (2)$$

Drag coefficient, C_D, was estimated from data available in NACA report 793[1]. Results of computations are plotted in Figure 1, where drive power versus speed is shown with rotor diameter as parameter. From the curves of Figure 1, one can immediately see the desirability of operation of the machine in vacuo to reduce drive requirements. Reduction of the ambient density, ρ, by a factor of 10 would reduce power by a like factor. Since such density reduction could be obtained with an ordinary mechanical vacuum pump, a decision was made to design the assembly for operation in vacuo. This decision had the immediate consequence of complicating design of support bearing since pressure seals would be required if high speed ball bearings were to be operated with conventional oil mist lubrication. Experience within this department includes high speed dry lubricated bearings which had been operated for many hours in pressure environments as low as 10^{-6} mm Hg, therefore a decision was made to employ special bearings designed for dry lubrication using MoS_2 as the lubricant. Fafnir Bearing people were consulted and agreed to supply a duplicate set of bearings having pure silver 1000-1200 Angstroms thick vacuum deposited on the balls. CBS laboratories agreed to vacuum deposit the needed silver at a cost of $350.00 but recommended a special coating of their own at a cost of one dollar per ball, or $13.00 per bearing. Upon the advice of the chief engineer at Fafnir, this CBS coating was selected for trial. With a source for special bearings established, the size of bearings fixed by speed, specific application of the bearings to a suitable

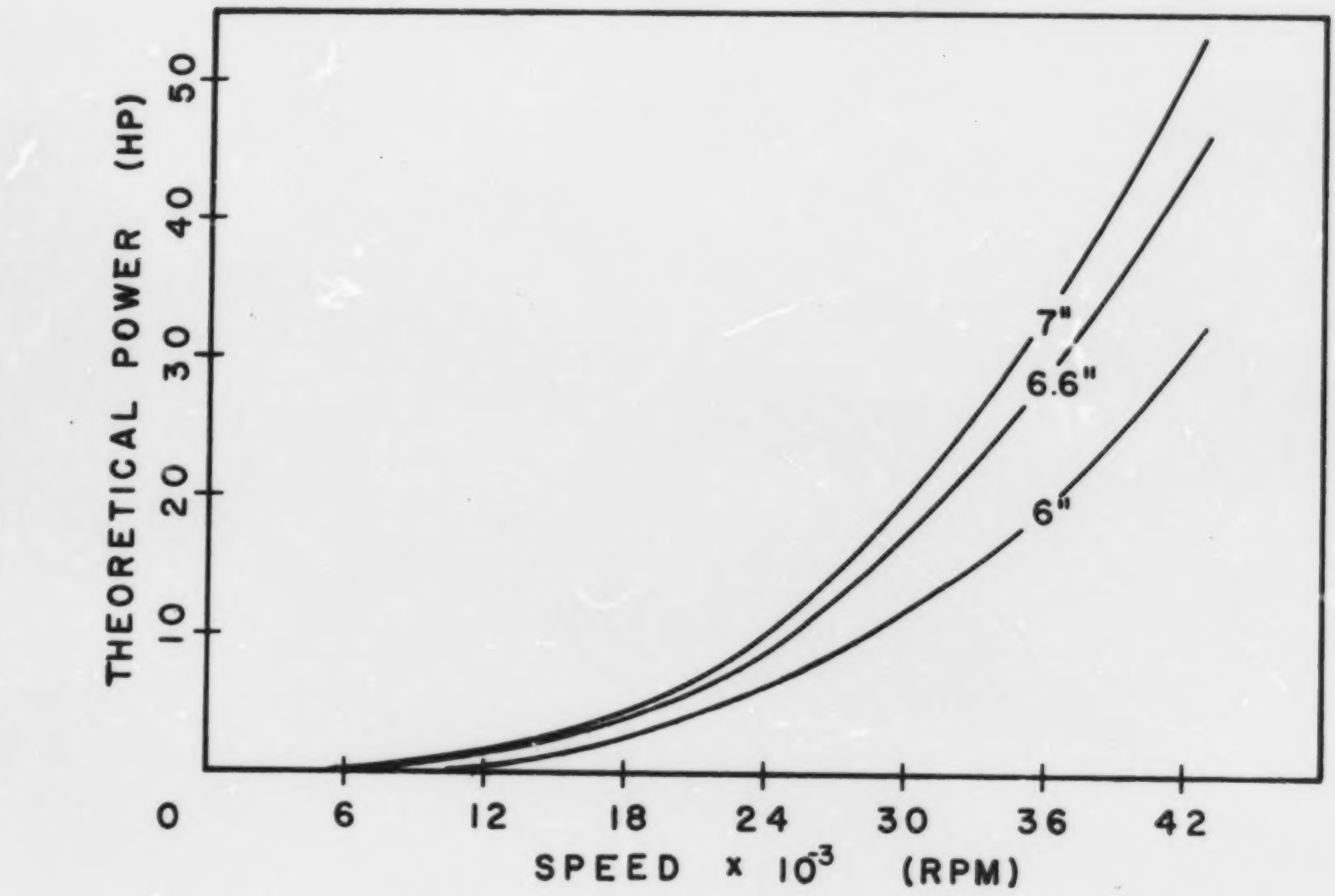
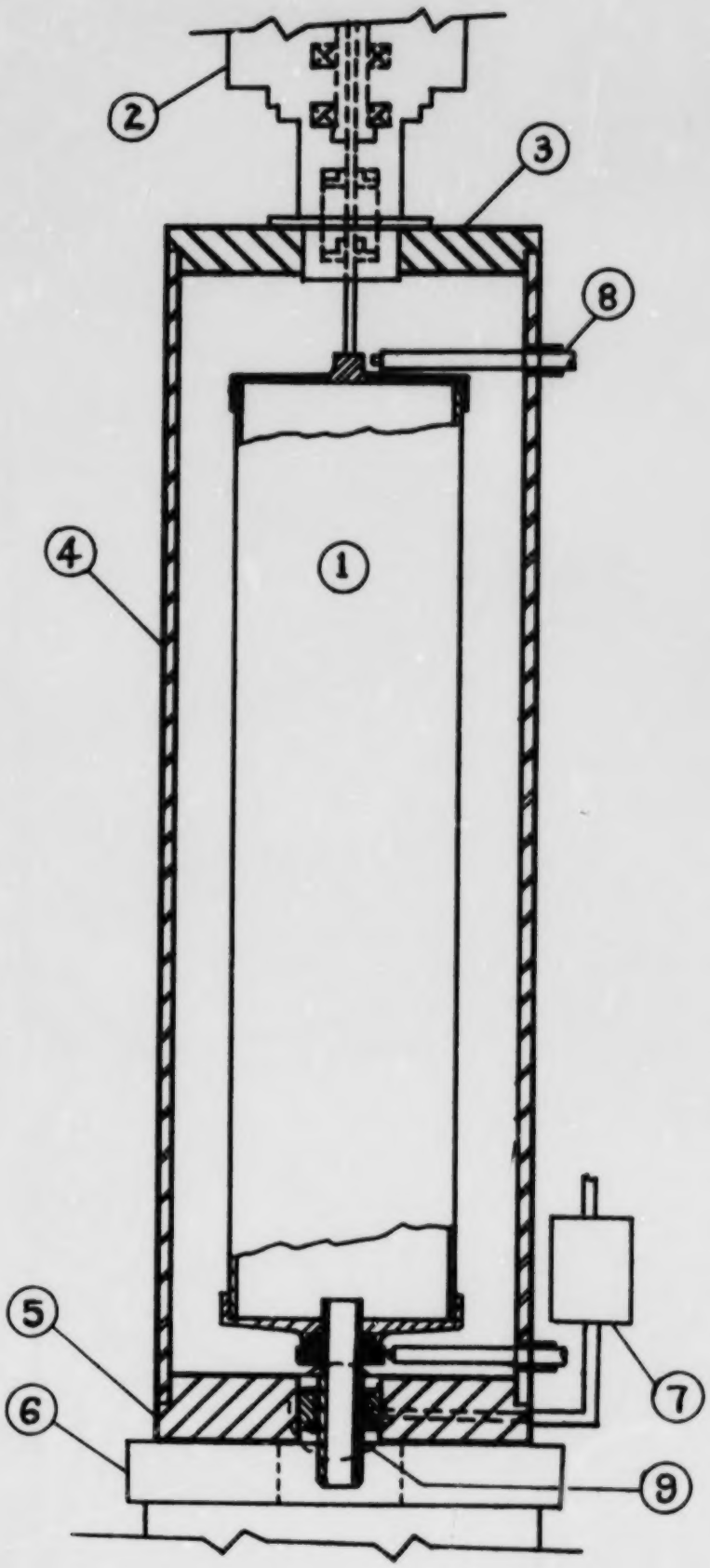


Figure 1. Theoretical Centrifuge Drive Power Versus Speed with Diameter as Parameter

support was undertaken.

Several alternate designs were considered which would incorporate ball bearings in rigid supports. Approximate rotor dynamic analysis raised a question of bearing life if supports were rigid, particularly if deposition of particulate were to produce a serious mass unblanace in the assembly. Further, the desirability of a drive line which did not require extremely precise alignment of bearings was considered necessary to an ultimate field machine. With these criteria, the machine was conceived as one vertically oriented within a jacket which would serve as both vacuum and safety mantle while providing for support of bearings. The throughput flow would be admitted at the bottom of the machine and would discharge into the vacuum jacket thus requiring ultimately, a vacuum pump large enough to handle the throughput flow as well as jacket leakage. Figure 2 is a cross sectional diagramatic view of the centrifuge assembly. The lower end cap - ball bearing - squeeze journal arrangement, item 9, Figure 2, is shown in somewhat greater detail in Figure 3. Since this design must be considered unorthodox, a brief discussion is in order. Referring to Figure 3, the stationary shaft provides for the inflow of sample and carries the non-rotating inner ring of the bearings. The bearings, modified Fafnir 3MM9105 ground for back to back mounting, were dry lubricated with MoS_2, the balls having been coated with a special vacuum deposited material, CBS coating #5940, MoS_2 was burnished into both inner and outer rings. These bearings are also non-standard as regards the retainers which were fabricated of "Rulon" rather than metal. Functionally, the stationary shaft is supported within the squeeze journal, oil for the journal provided by an external reservoir to insure complete flooding. Journal clearance was set initially at 10 mils. which was considered



ITEM NO.	DESCRIPTION
1	Rotor
2	Turbine
3	Top Plate
4	Jacket
5	Bottom Plate
6	Lathe Bed Block
7	Oil Reservoir
8	Bently Probe
9	Squeeze Film Bearing Assembly

Figure 2. Diagramatic Section of the Centrifuge Assembly

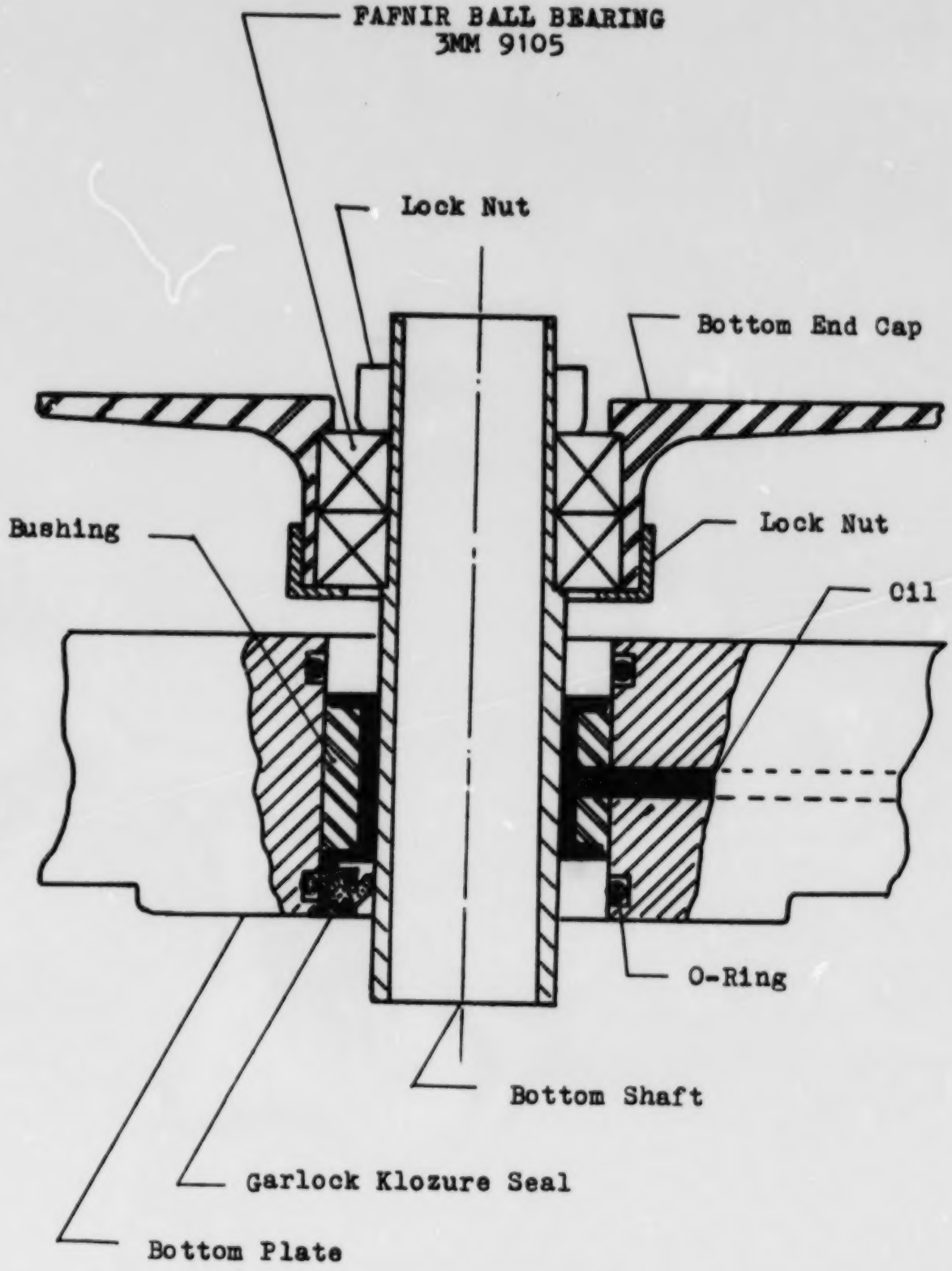


Figure 3. Section View of Rotor Lower Assembly and Squeeze Film Bearing Support

adequate to permit steady lower shaft displacement (run out) as large as 5 mils. This clearance also provided for run out due to slight misalignment of the drive line.

The upper bearing support for the centrifuge takes advantage of a new drive turbine, available commercially from Barbour Stockwell Corp., in that the turbine bearings, both support and damping, are employed rather than a second bearing assembly for the centrifuge shaft. This turbine was contributed to this work by the School of Engineering and Applied Science, at cost of 2500.00,to replace a less suitable existing in house machine.

The design work outlined was completed during May of 1968 and working drawings submitted to the in house RLES Machine Shop. Completion was expected early July. Delays in receipt of some components particularly the special bearings held up the start of mechanical evaluation until September. Following the completion of shop drawings, attention was shifted to design of a suitable laboratory aerosol system and associated hardware necessary to permit centrifuge induction from the system. Since the external aerosol system had to be sized based on throughput, the analysis employed to achieve throughput control follows.

Throughput Control

Ultracentrifuges were known to be self pumping as was indicated in the original proposal for this work. This capability arises from the radial pressure gradient produced by the high speed rotation coupled with either suitably positioned orifices or internal 'scoops' which take advantage of the available total head within the machine. Orifices located in the outlet (upper) end cap were decided upon and the number,

size and location had to be decided. A first approximation of interal radial pressure gradient was made as follows: Consider the throughput to be an ideal gas rotating about a vertical axis with angular speed ω, rad/sec. The pressure is expressed by

$$dp = m\omega^2 \rho a da \tag{3}$$

where m = mass of the molecule

ρ = molecular number density

a = radius

From kinetic theory, pressure and number density are related by (assuming uniform temperature)

$$p = \rho kT$$

from which

$$dp = kT d\rho \tag{4}$$

Combining equations 3 and 4

$$\frac{d\rho}{\rho} = \frac{m\omega^2 a da}{kT} \tag{5}$$

Equation (5) with boundary condition

$$\rho = \rho_o \text{ at } a = a_o$$

gives

$$\rho = \rho_o e^{m\omega^2 a^2/2kT} \tag{6}$$

or in terms of pressure

$$p/p_o = e^{m\omega^2 a^2/2kT} \tag{7}$$

Equation (7) was employed to calculate values of the ratio p/p_o for various ω in the range $0 \leq \omega \leq 800$ rad/sec. Results of these computations are shown plotted in Figure 4 as p/p_o vs radius with ω as parameter. With these gradients, computation of through put was possible by selecting a radius and then establishing the pressure ratio which would exist between the inner environment and the exterior ambient. This was considered more tedious than desired and an alternate procedure was contrived. Considering the desired throughput of 0.5 m^3/min, the area required for flow with different Δp could be established. This approach enabled plotting curves of numbers of orifices required versus orifice diameter with Δp as parameter. Since Δp in turn depended on radius for a given speed, effect of speed changes for a given choice could be approximated. Information provided by Figure 5 resulted in the choice of 8-0.125" orifices to be equally spaced on a 4.500 inch hole circle on the outlet end cap. For conditions anticipated, throughput flow was expected to vary linearly with speed since the exponential of equation (7) relates p to the square of the speed and the flow relationship for orifice flow involves square root of pressure difference. This linear relationship was realized as will be seen in a later discussion of experimental results. Clearly then, throughput control required either change in the number of orifices employed at a particular speed or for a given choice of orifices, change of speed. A third possibility would be alteration of the centrifuge core pressure (p_o, of equation 7) by some type of flow restriction located in the inlet arrangement.

The overall laboratory aerosol system was constructed with advice, regarding various methods of producing aerosols with known particle sizes, from Dr. Jack Wagman and his staff at NCPAC, Cincinatti, Ohio. Based on

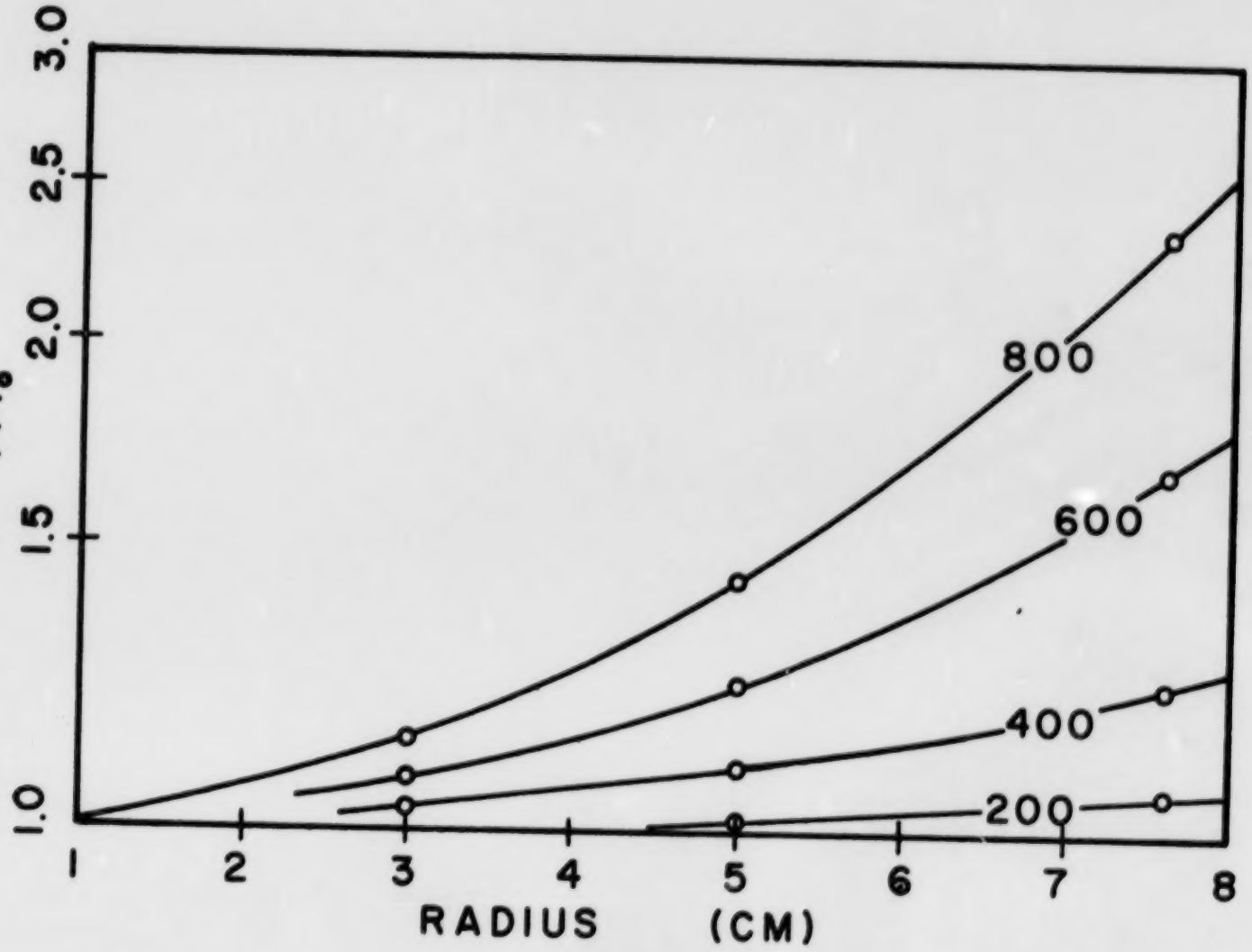


Figure 4. Radial Pressure Versus Radius with Angular Speed as Parameter

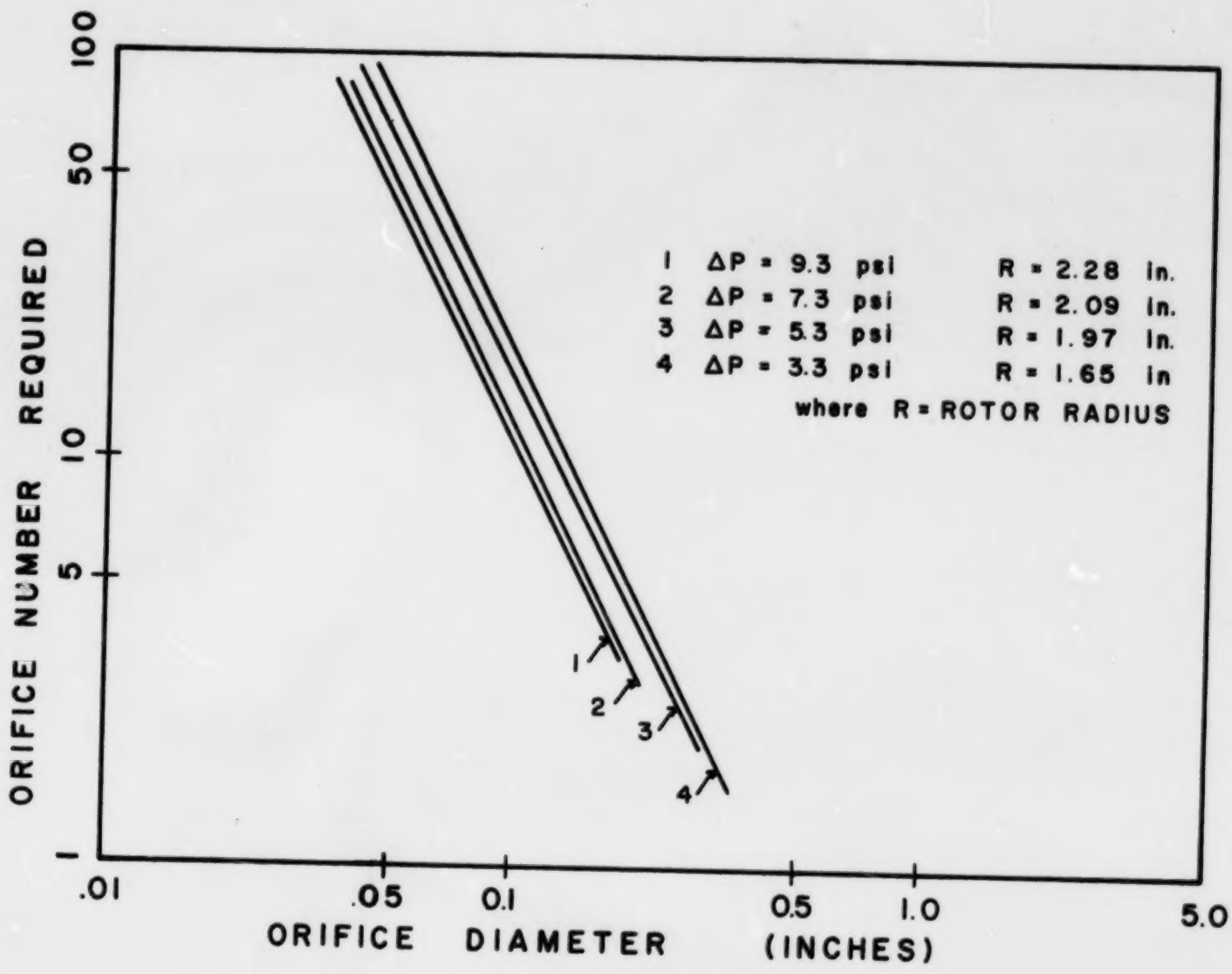


Figure 5. Orifice Size and Number Selection Chart

this advice, a Dautrebande model 95 generator was arranged to discharge into a large volume hold up chamber from which the centrifuge induction would occur. The resulting arrangement is shown schematically in Figure 6. The hold up chamber was sized large enough to provide time for the drying of particulates generated by the Dautrebande. The second chamber, located at the rotor inlet, was sized to approximate induction from an infinite environment. This system was essentially completed and ready for operation in September 1968, through because of unexpectedly difficult mechanical problems which were encountered, was not put to good use until March 1969.

All mechanical components were completed during July 1968 and with the arrival of the special bearings on August 23 initial assembly was made. The centrifuge drive shaft was press fitted into the upper end cap, a 0.010" interference fit employed to insure differential radial expansion induced at high speed and thermal expansion due to viscous drag heating would not separate the Al 7075-T6 alloy end cap from the steel shaft.

The inner-outer ring fit specified for the special bearing was 0.0005" tolerance. These bearings could not be assembled by tapping or application of even light force because of the serious mechanical problems even small blows can cause. The lower shaft was found to have small surface imperfections which prevented forceless assembly and a new shaft had to be ground. To insure a no force assembly, this shaft was chilled in liquid Nitrogen and with the aid of a small alignment jig, the bearing inner rings were slipped on. The alignment jig was necessary to prevent cocking of the rings which could have caused them to hang. Had they done so, it might have proven to be impossible to remove or properly seat them. With the rings in place, the balls, "Rulon" retainers and outer rings were installed the inner ring

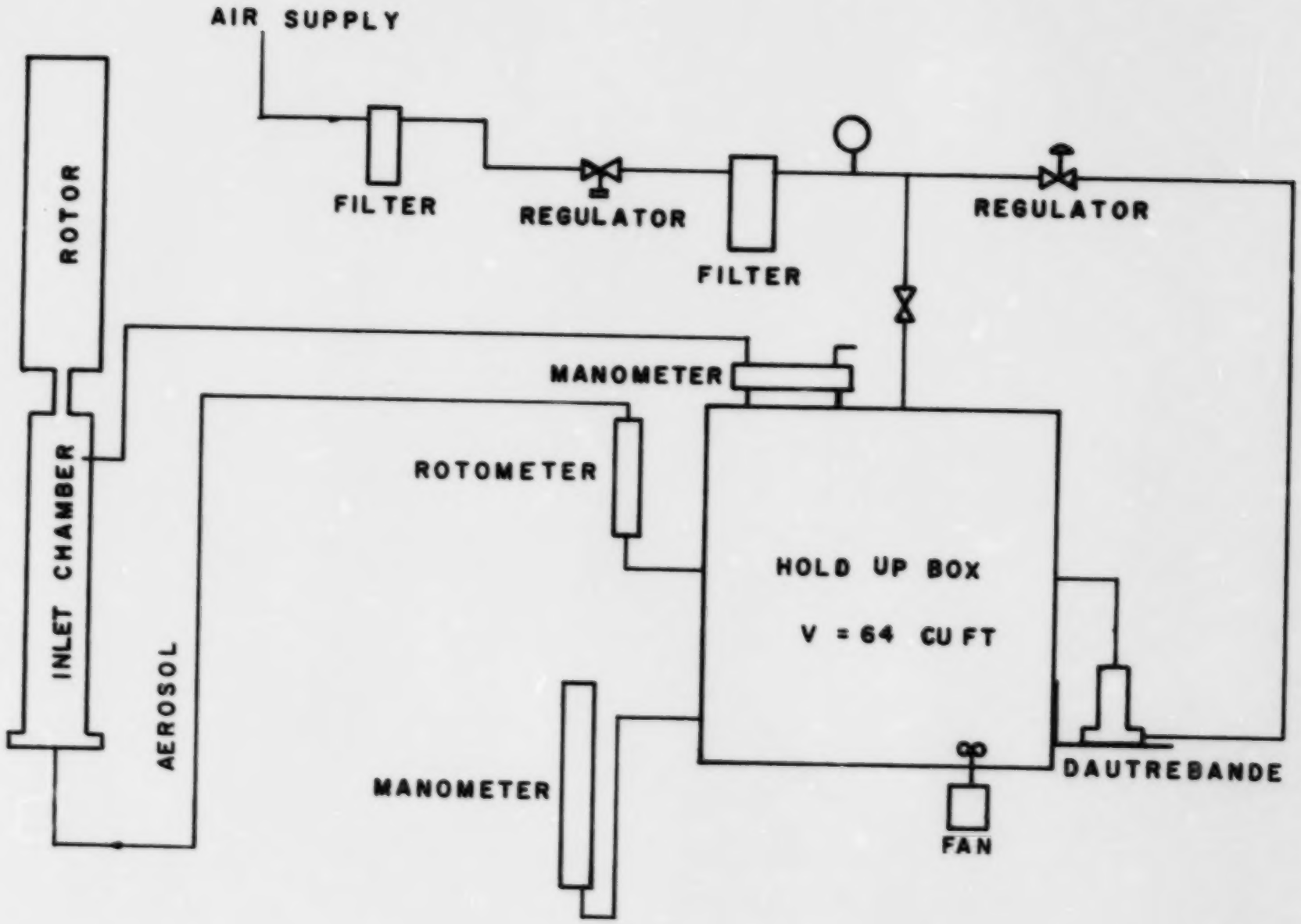


Figure 6. Schematic Diagram of Aerosol System

lock nut then placed and torqued. This sub assembly was then fitted into the Al-7075-T6 alloy inlet end cap bearing well, the aluminum piece having been heated to 180°F. No difficulty was encountered here and the aluminum outer ring clamp nut was torqued down. The end cap sub-assembly was then ready for assembly to the centrifuge tube. Figure 7 is a photograph of the several component parts of the machine excluding the special bearings.

Figure 7. Centrifuge Components

EXPERIMENTAL PROGRAM AND RESULTS

Experimental Arrangement

The interrelation of physical components is illustrated schematically by Figure 8. The arrangement shown was largely fixed by fact that the centrifuge, centrifuge drive, and the vacuum jacket provided an integral assembly rather easily mounted in high speed rotor test facility already available in house. External connections required were those associated with supply of air to the drive turbine and lubricant to the several bearings. A modification of the Barbour-Stockwell turbine damping bearing was made to permit non-circulating pressurized feed of lubricant, this change considered desirable as a means of varying upper spindle damping slightly.

Mechanical design evaluation was originally thought to be possible as a by-product of experiments directed to study of internal gas flow. Based on this approach, the original dynamic instrumentation included a pair of Bently Proximity probes, arranged to measure upper and lower spindle displacements. These probes, with associated circuity shown diagramatically in Figure 9, produce a reasonably linear, relatively large voltage variation with displacement. Typically, a one mil change of spindle position produces an output change of two volts, making detection of small fluctuations possible. Serious mechanical trouble encountered during initial operation required addition of four such probe stations. Two were added 90° ahead of the original pair and a similar pairing was added at rotor mid plane.

Speed measurement for all runs was provided by a magnetic pickup located on the upper plane of the turbine wheel. Pulses produced by cylindrical cavities on the wheel, 6 in number per revolution, were counted

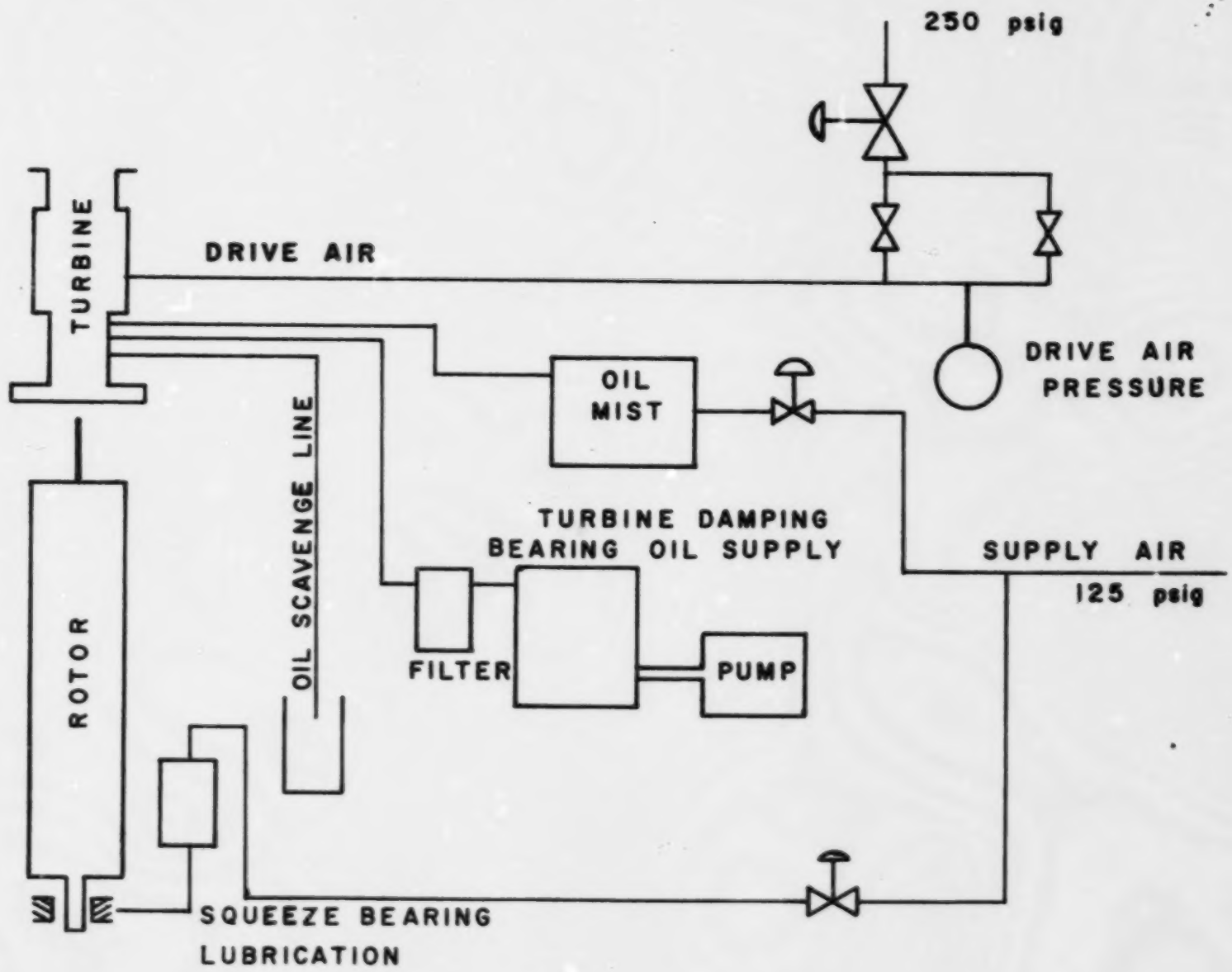


Figure 8. Schematic Diagram of Centrifuge and Drive

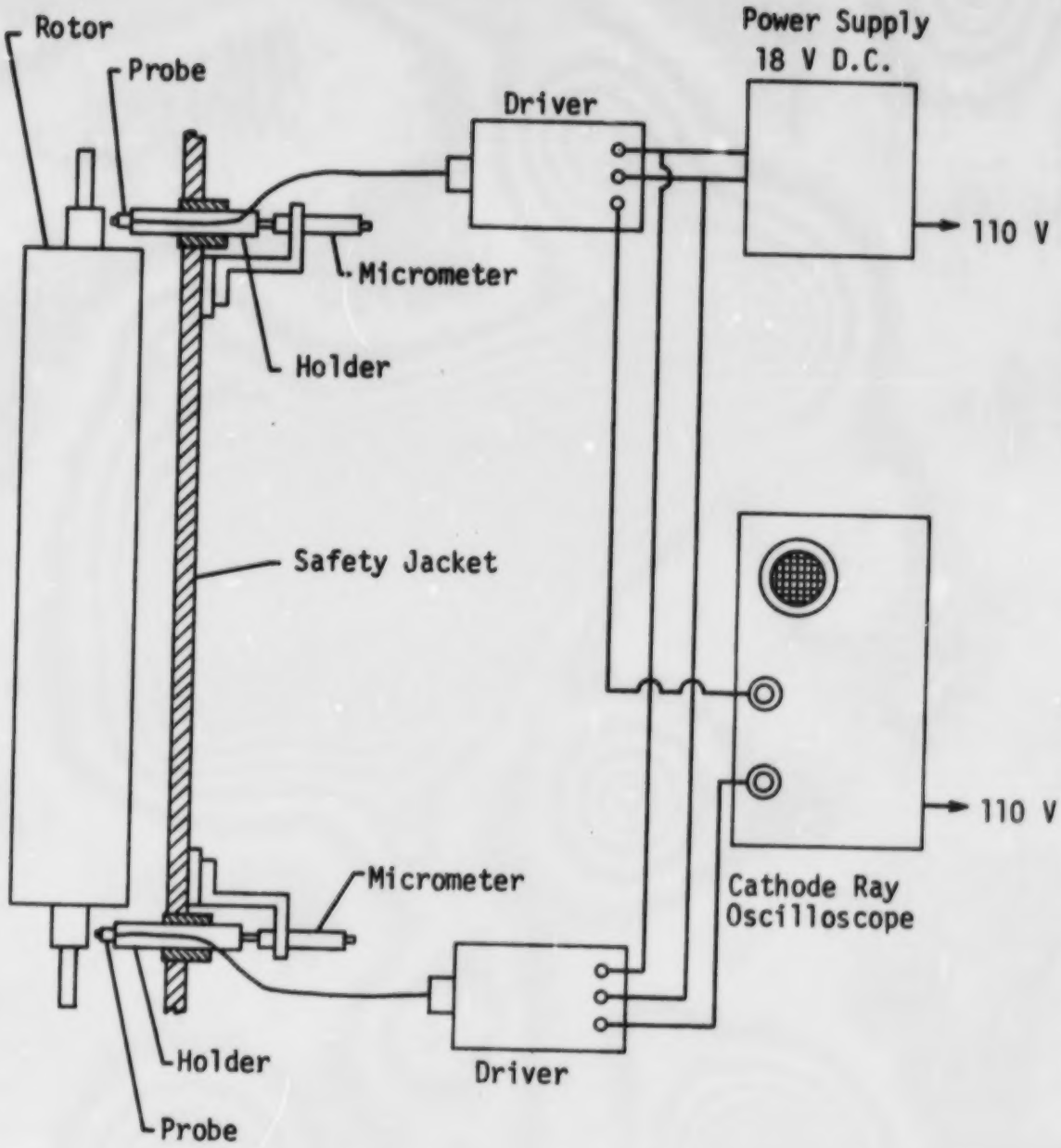


Figure 9. Schematic Diagram of Bently Probe System

with a Hewlett-Packard model counter. A second speed measurement was made using the frequency of the output sine wave produced by the proximity probes. A third light source-photo multiplier arrangement was added later to provide both an additional instantaneous speed measurement and a source of an external triggering signal for the probe read out oscilloscope. Figure 10 is a photograph of the centrifuge assembly mounted in the test facility, the associated external equipment arranged as shown in Figure 11.

Mechanical Design Evaluation

A complete mechanical assembly of the centrifuge was put into operation early in September 1968 and was quickly found to have serious mechanical problems. Lower end spindle run out exceeded the 8 mils clearance provided by the squeeze journal and comparison of upper and low proximity probe read out suggested an out of phase motion of the upper end spindle relative to the lower. Particularly serious was the very low speed, 2000-4000 rpm, at which this occurred. The out of phase relationship of the probe pair suggested a different precessional motion was present at each machine end spindle. Such a motion was considered detectable using a pair of co-planar probes positioned 90° apart. This arrangement with the probe signal of one of the pair fed to the x, the other to the y coordinate of a dual beam oscilloscope, such as the Tektronix 502 used here, provides a direct display of spindle precessional orbit. Additional probe stations previously described were added to secure this type of data readout.

A careful study of the signals realized with this new arrangement of probes disclosed a rocking motion of the stationary inlet shaft within the double bearing assembly. Such a motion was only possible if the locking provisions failed to sufficiently pre-load the bearings since rigidity depends on this pre-load. Correction required grinding a new shaft, the

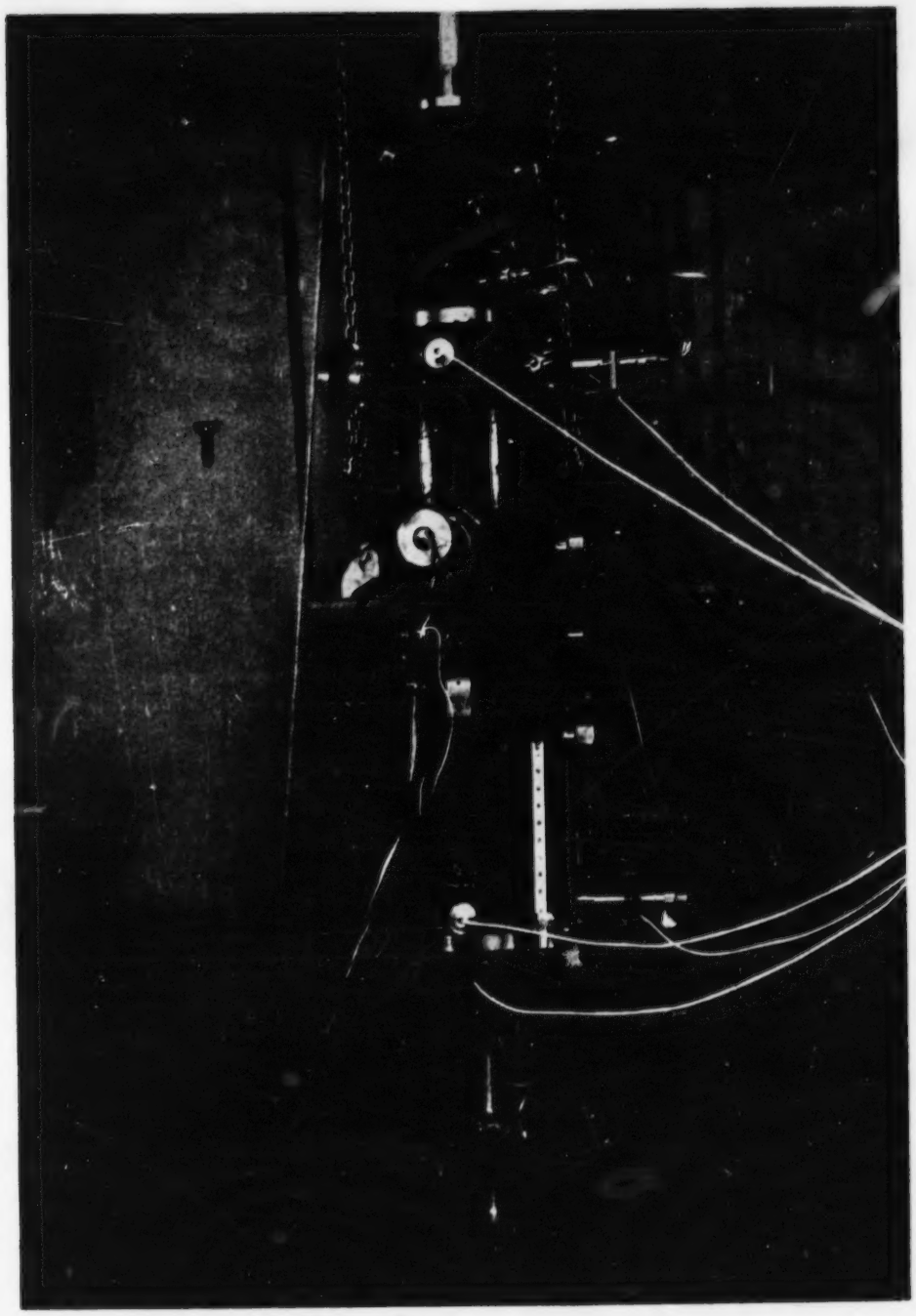


Figure 10. Installed Centrifuge Assembly

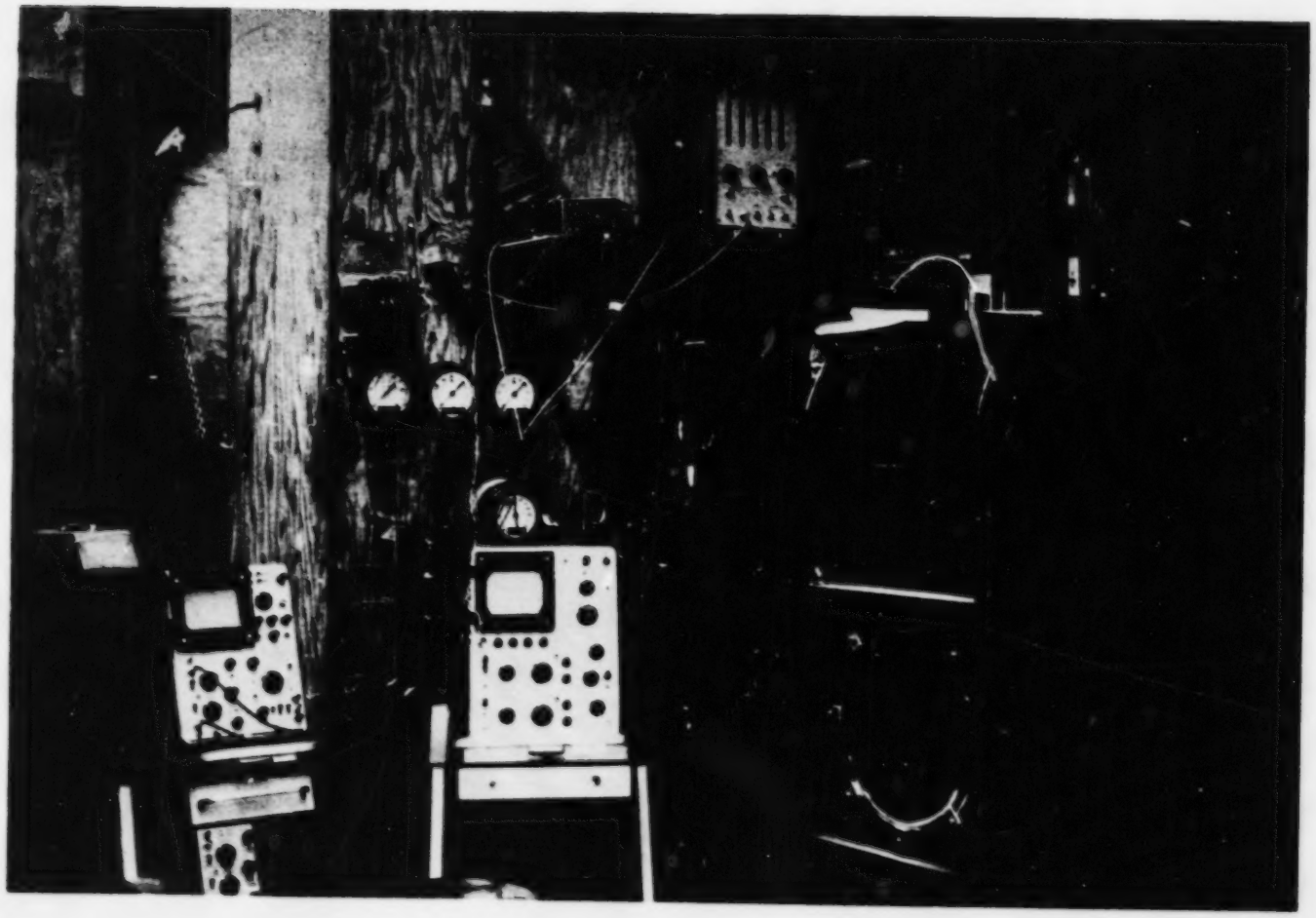


Figure 11. External Experimental Equipment and Instruments

third, since the difficulty was traced to the inner ring stop shoulder not square to the longitudinal axis. Operation with this new shaft appeared to have removed the rocking action, but lower spindle runouts to 6 mils were present at 4000 rpm. Probe data again provided information pointing to the trouble, in this case the pair of probes added at mid plane. The probe signals exhibited a frequency twice that corresponding to rotor speed. Such an occurrence would only be possible if a non-circular section was rotating past the probe. Since the tube had been carefully mapped upon its completion and found to be round within 0.0001" inch, this seemed an unlikely physical cause and circuitry became suspect. No electronic problem was found, so the machine was again removed from the experiment, fitted to a spindle support jig and remapped. This survey revealed a skew distortion had occurred approximately midway between end spindle support bearings. Physically, such a distortion, 8 mils maximum in this case, creates a mass unbalance which in turn causes the rotational axis to be displaced from the geometric when operating at speed. The squeeze-film journal had been designed to permit the shaft to find whatever new rotational axis necessary without producing destructive bearing forces as would occur in a rigidly supported bearing. In this case, the unbalance was large enough to cause runout in excess of that which could be accommodated by the squeeze journal. The work was now well into October and a decision regarding production of a new tube had to be made. Corrective balancing could only be attempted with the machine in its own support bearings and for a machine with damping this procedure is problematical. What was to be done required consideration of the probable cause of the distortion and pursuant to the reasoning involved it is necessary to outline the machining process required to produce the tube. Raw material was an extruded pierced billet of Al-7075-T6 alloy, 7 1/2" OD with a 5/8" wall. This was mounted in a 27" swing engine

lathe and the inner diameter carefully machined to 6.500". With this bore established a prepared mandrel of 0.001" larger diameter was chilled in liquid notrogen and the material slipped on. The mandrel was then mounted into the lathe and machining of the outer diameter followed. When size approached desired final dimensions, the material was heat treated by quenching in liquid nitrogen followed by immersion in boiling water with subsequent cooling in air. This was done four separate times, between four finish cuts of less than 0.001", each in an attempt to relieve all residual stresses caused by machining. Despite this careful heat treatment, residual stresses apparently were present and were relieved during initial operation of the assembled machine. Since production of this first tube alone had taken 110 hours of machinist time at a cost of approximately $1,000.00, machining a new tube appeared to be an expensive risk offering no assurance the same trouble would not occur. Further, machining a new tube would probably require producing new end caps as well since the tube diameter was established first and end caps then carefully fitted. Working in reverse would be a difficult task for even the most competent machinist. The task of in-line balancing appeared to be the more realistic route to follow and the decision was so taken. The method selected was that described by Thearle [2]. Application proved every bit as tedious as was anticipated since three runs were needed to obtain even a first trial set of balance masses. Successive additional runs were then made, each new run supposedly leading to more accurate balance mass position and quantity. The technique amounts to a kind of experimental iteration and did produce considerable improvement in dynamic behavior allowing speeds to 8000 rpm. The method failed to provide improvement above this speed however, which raised a question concerning the adequacy of the squeeze

journal design. A major modification of the entire lower end cap-bearing-squeeze journal arrangement appeared necessary. So drastic a step with the attendant loss of time and probable cost of new hardware made necessary finding some alternative. This writers colleague, Dr. Edgar J. Gunter, an expert in the area of high speed rotor dynamics, was consulted. Dr. Gunter has been engaged for some time, in an effort to secure solutions of the complex equations of motion for rotor-rotor systems. This work had produced a computer program, which with modification, appeared applicable to the ultracentrifuge. This was done and while first results did not immediately remove the troubles, the trends predicted correlated well enough with experimental measurements to suggest the support assembly was not at fault. Continued refinement of the program as dictated by experimental results ultimately predicted needed balance masses and location sufficiently accurate to essentially remove the trouble. Rotor speeds to 20,000 rpm were realized, and operable limits of permissible run out established. The experience accumulated also provided ready reference information whereby any peculiar operation could be evaluated as to seriousness during a run. The details of this computer work will be deferred to the final report of this centrifuge work, this deferment due in part to the rather lengthy display of equations necessary and in part due to several as yet incomplete but potentially valuable correlations realized as by products. These include correlation of bearing forces with speed, damping and its effect on both run out and force transmitted to bearings, amplitude and phase angle versus speed information. This phase of the work will be published in the open literature, with due credit to NAPCA as sponsor, when better correlation has been realized.

The value of this computer analysis can be outlined briefly by a comparison of computer predicted relationships with those realized

experimentally. Figure 12 is a copy of the computer print out of amplitude versus frequency. The two curves marked by □ symbol represent the centrifuge operating with 5W lube oil in the squeeze journal. The solid lines depict lower spindle motion, while the upper spindle is represented by the broken lines. Curve marked X and Z are predicted displacements with more viscous oil in the journal, physically representative of increased damping. The measured amplitudes are shown plotted versus speed in Figure 13. Comparison quickly reveals motion of the upper spindle to be quite similar. The cylindrical critical occurred very nearly at the predicted speed of 3500 rpm. The lower spindle did not go through a larger displacement, the conical mode, at 6000 rpm, but it did suffer a small increase of amplitude at 5600 rpm. A similar comparison of predicted phase angle versus speed, Figure 14 with experimental results, Figure 15 reveals a less valid correlation. However, the general curve form is quite similar and the magnitude, in degrees, of the phase angle predicted between 1000 and 4000 rpm was in quite good agreement for the upper spindle. The computer predicted bearing forces versus frequency, Figure 16 reveals significant information, though unfortunately is a measurement which cannot be accomplished. Note that the bearing force decreases at the upper spindle, the trend essentially following the amplitude curve. Even the non-correlated lower curve is useful for it exhibits a minimum at 8000 rpm. More interesting is the force predictions for cases x and z. Referring back to Figure 12, these two cases computed for increased damping indicate the increase will decrease amplitude, that is, reduce the run out. While this at first glance appears desirable, examination of the force curves in Figure 16 indicate bearing forces are increased, those predicted for the upper bearings quite sharply. One experimental run with heavier

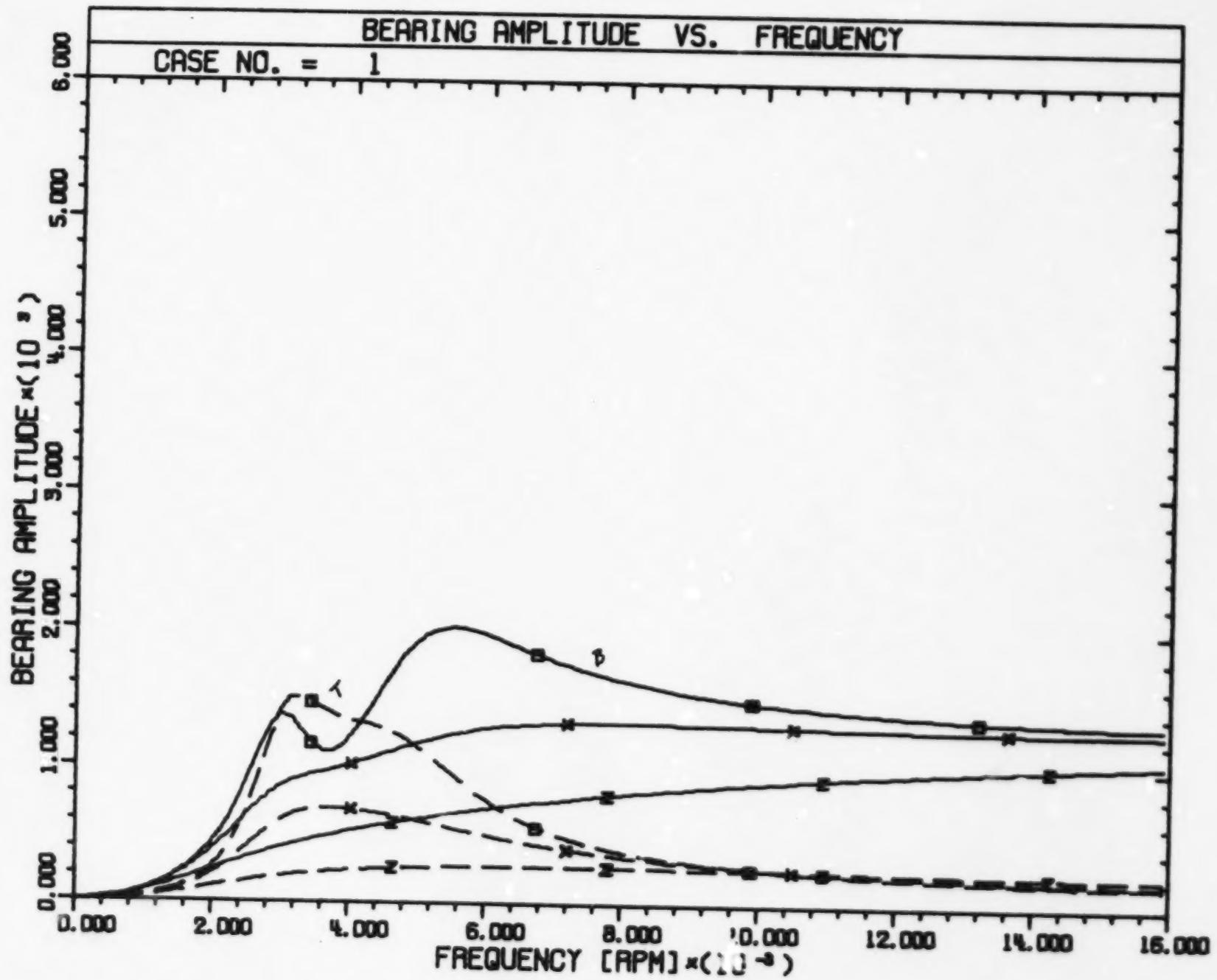


Figure 12. Computer Predicted Amplitude Versus Speed

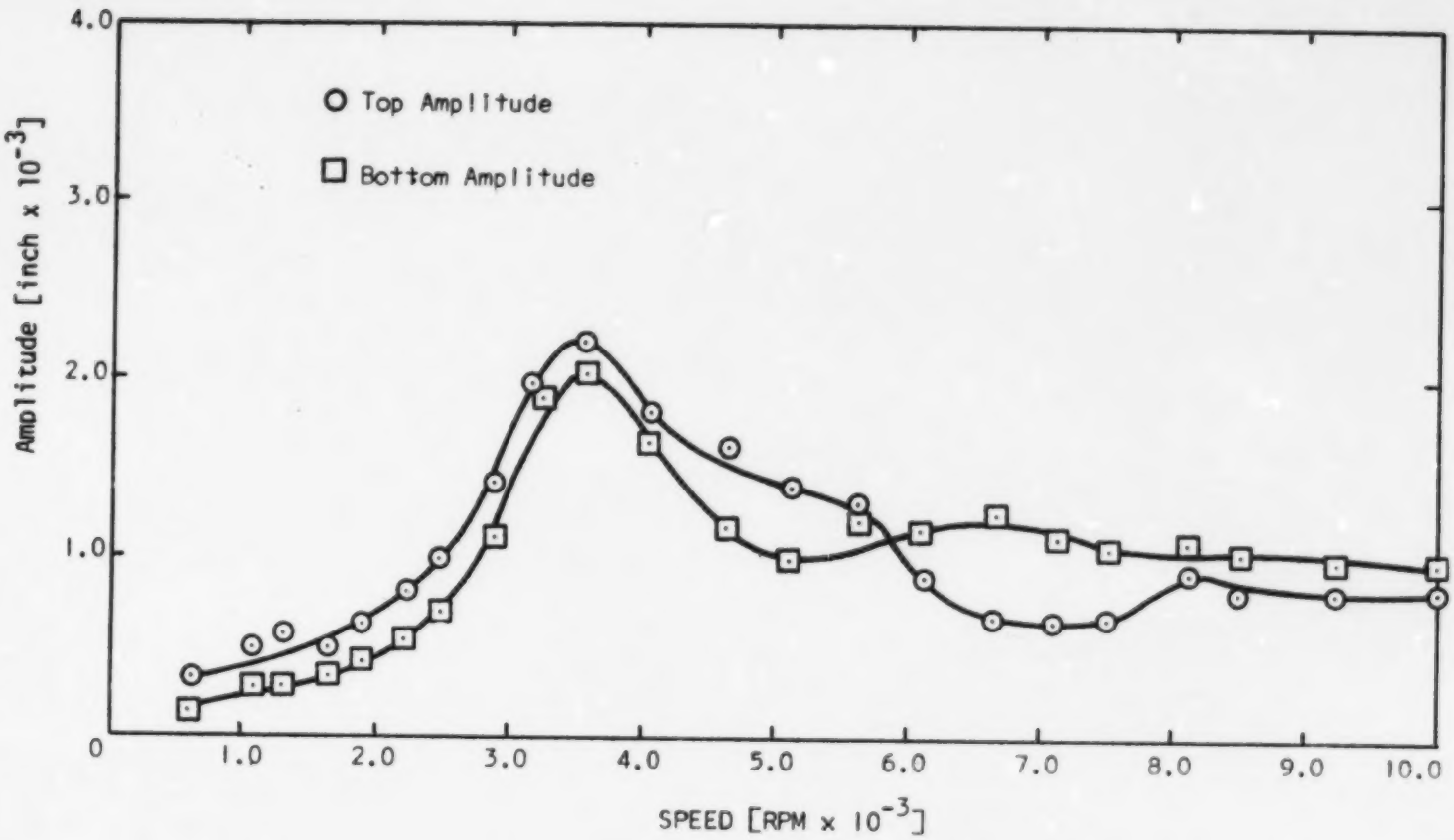


Figure 13. Experimental Amplitude Versus Speed

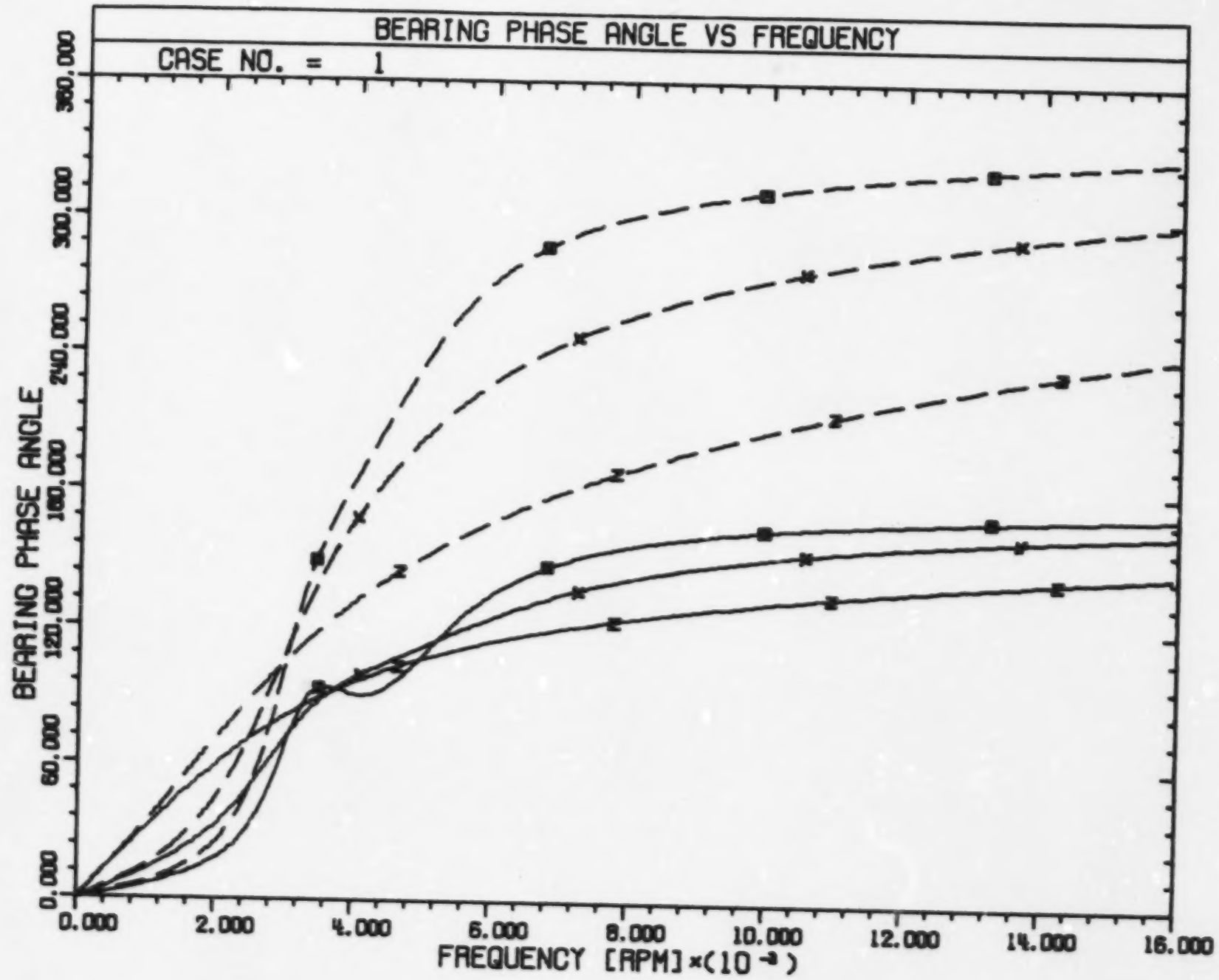


Figure 14. Computer Predicted Phase Angle Versus Speed

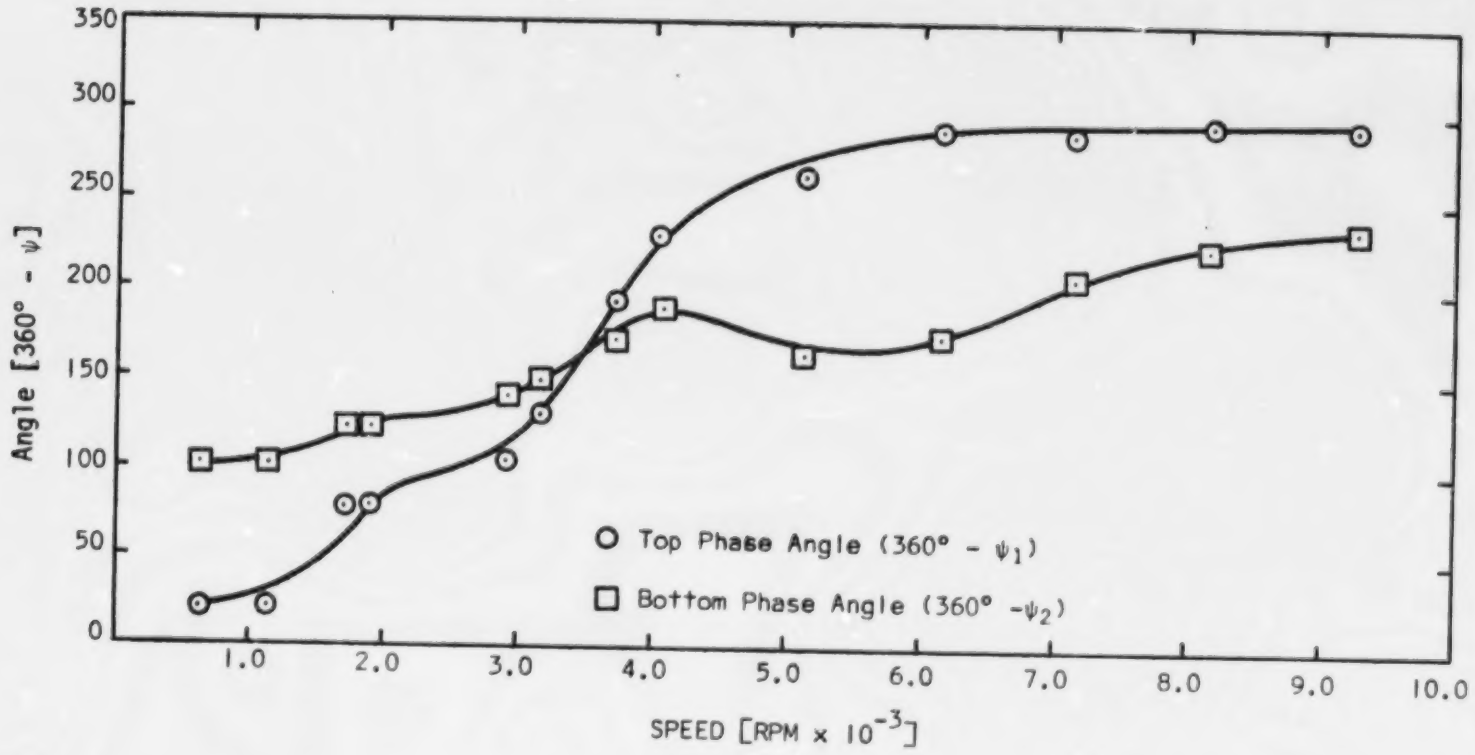


Figure 15. Experimental Phase Angle Versus Speed

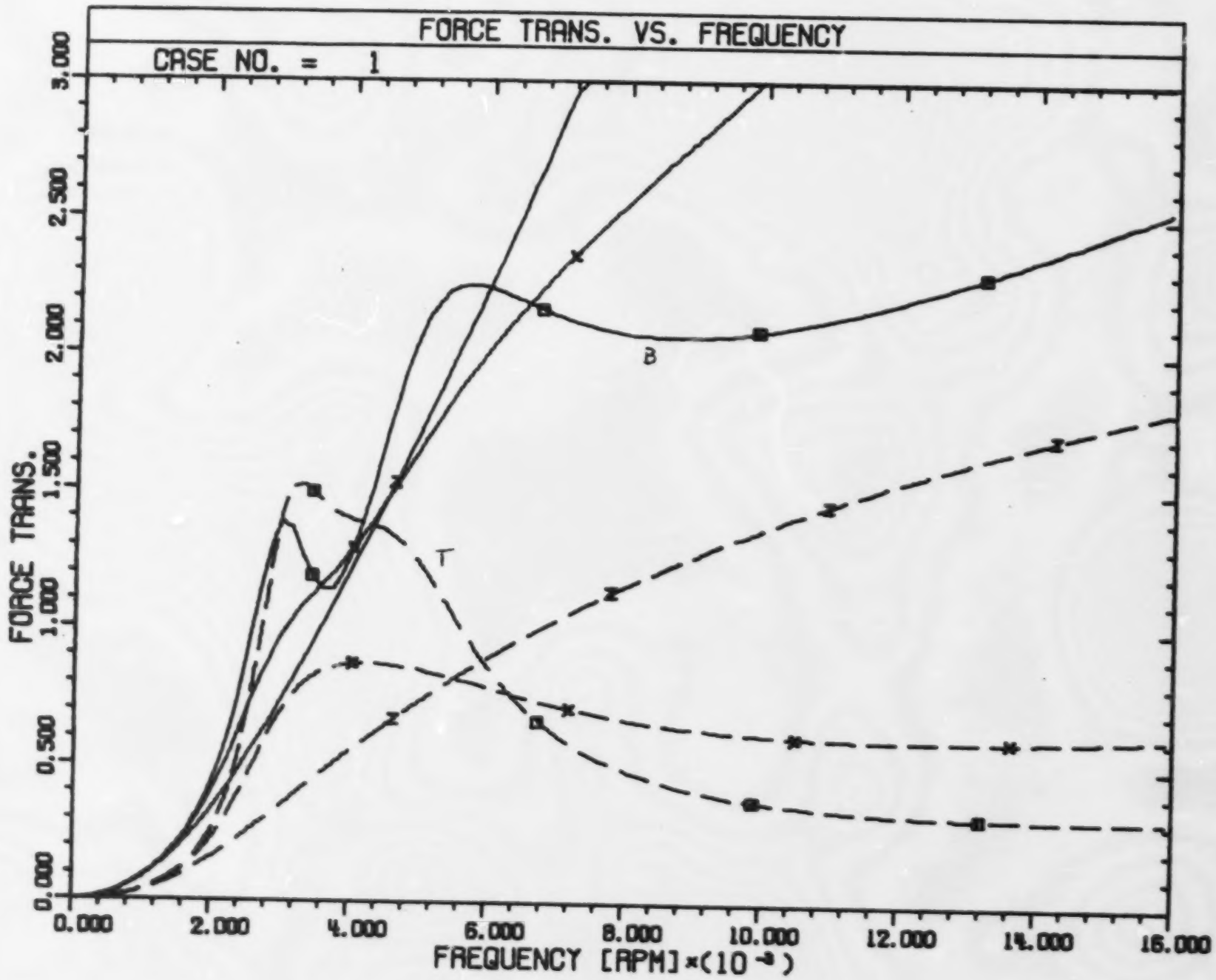


Figure 16. Computer Predicted Bearing Force Versus Speed

viscosity oil was made but the results were not particularly conclusive. Further work with this computer analysis of performance was deferred when the centrifuge operation provided high enough speed to permit a serious start on flow pattern work. Speed of 8000 to 12000 rpm were estimated to be high enough to permit work with particles as small as 0.25μ and attention was turned to the fluid mechanics work in March 1969.

INTERNAL FLOW STUDY

The very approximate model originally outlined in the proposal upon which this contract work was based established only the need of a Stokes flow law regime within the machine. Further analysis did not materially improve a priori knowledge but suggested particles of a given size should appear in bands spaced non-linearly along the bowl surface. It was anticipated the inlet geometry would play a large role in achievement or failure to achieve particle classification. Since an analytical solution was not possible, experimental shapes and their influence had to be studied.

Integrally involved with this flow study was the method whereby particulate could be collected. A removable lining was the obvious choice , but how to install this lining, remove it and then examine it for particulate deposit pattern were questions to be answered. Critical at this juncture was the possible serious dynamic problems which could attend unbalance caused by the sample collection lining. This latter problem was found to be non-existent. Several lining materials were employed usually installed as full internal surface cover. Materials included high quality vellum, mylar film, plastic coated paper and vinyl plastic film. Before discussing deposit pattern work further, results of throughput versus speed measurements will be presented. The reader will recall the method whereby the outlet orifices were sized and located. All flow runs performed during this report period were made with the outlet end cap having 8-0.125" orifices equally spaced on a 4.500" hole circle. Figure 17 is the volume flow versus speed relationship as determined by measurement. The inlet geometry for this run was simply the open stub inner end of the lower spindle. Other shapes to be described shortly did not markedly alter the flow relationship shown, suggesting the control lay entirely with the outlet orifice arrangement.

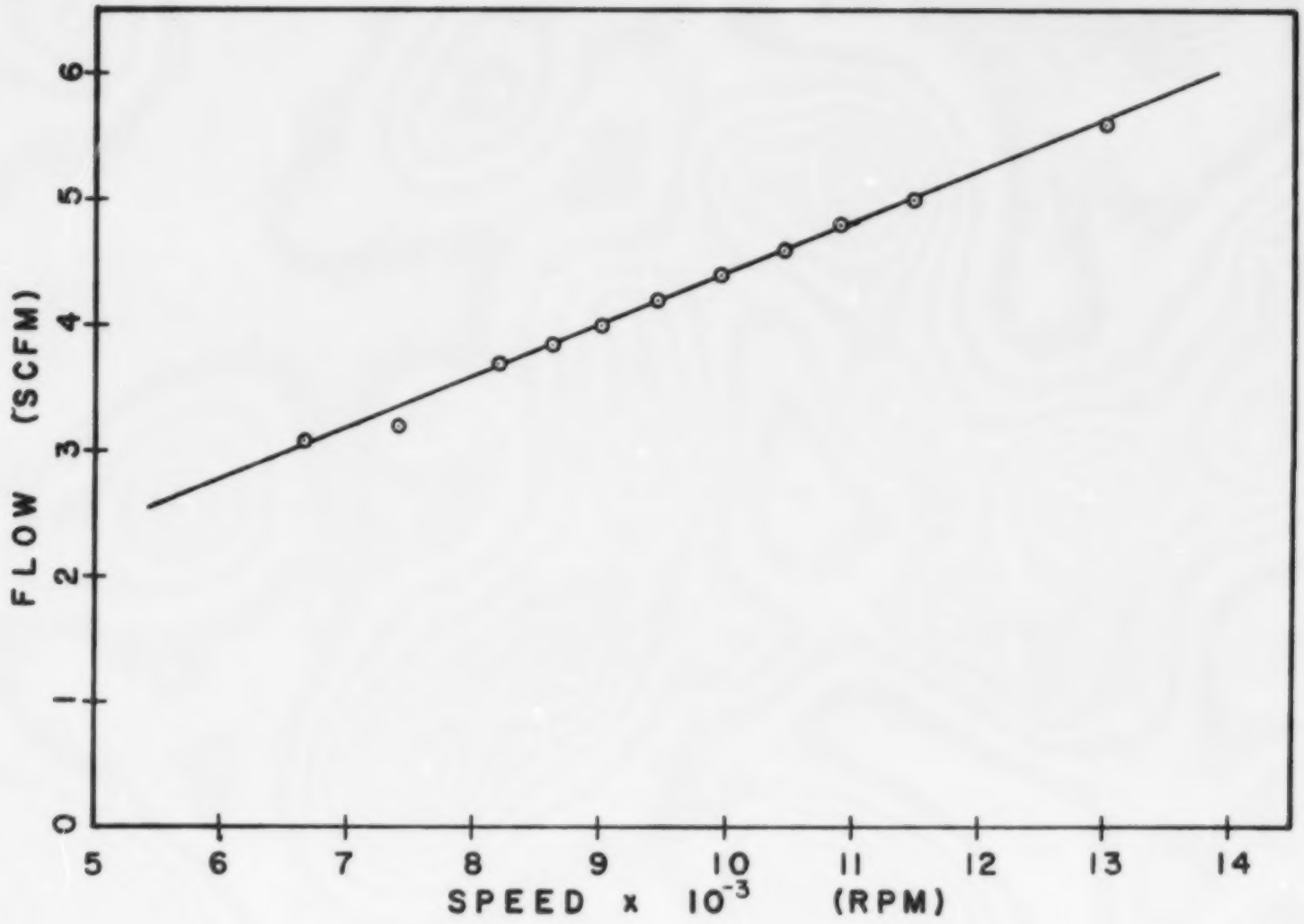


Figure 17. Throughput Volume Versus Speed
Outlet 8-0.125 Orifices Equally Spaced on 4.500 Circle

While the flow-speed relationship shown is self-explanatory, two things are noteworthy. (1) the measured flow is linear with rotor speed as expected for this speed range and (2) the flow values are within 10% of those predicted. One unexpected result not self evident was the onset of a flow pulsation first noted at a speed of 13,000 rpm, and evidenced by an oscillation of the fluid in the manometer sensing inlet chamber pressure.

Oscillation frequency and amplitude increased with increased speed, becoming very severe at 16,000 rpm and speed was reduced to 12,000 rpm. Cause of this behavior was initially taken to be a choked flow condition at theoutlet orifices, but detailed flow computation ruled out this possibility. The precise cause has yet to be determined since a reasonably valid internal flow model has not been established. A possible explanation is a possible fluctuation of some type of secondary flow present in the region of entrainment which gives rise to an instability. The puzzling aspect of the behavior is its periodicity when all other variables associated with throughput appear to be time invarient. Clearly this pulsation can not be tolerated. Discovery of this behavior was not crippling since the throughput rate of 3 cfm and a speed of 10,000 rpm was sufficient to warrant serious particulate classification work.

AEROSOL EXPERIMENTS

Aerosol throughput experiments were initiated during the first week of April, the sample collection surface a high grade Vellum lining which covered all of the inner surface of the machine. This removable material had been installed during some of the latter stages of rotor balancing work in order that its influence on the dynamic behavior might be evaluated. The centrifuge inlet was simply the open bore of the lower shaft. Connection of the rotor to the aerosol system was accomplished with an especially machined adaptor, an earlier flexible hose arrangement having proven unsatisfactory. Recalling here, again, the design function of the lower support assembly squeeze journal, direct solid connection of any external tubing would defeat its purpose of providing a limited motion of the shaft.

Particulates of known size were provided by Dow-Chemical in the form of a monodisperse suspension of polystyrene latex spheres in water. Generation rate was established, approximately, through dilution of this material based on the Dautrebande No. 95 generator characteristics, the desired particle product rate intended to yield a concentration of 10^{10} particles/m^3. With available optical microscopy limited to 1000x magnifection, 0.81μ polystyrene latex spheres were selected for initial deposit study. A procedure was developed which permitted bringing the centrifuge to a desired operating speed, allowing the machine to pump clean dry air from the aerosol system during run up. The small mass of the assembled rotor, slightly under 8 pounds, permits relatively rapid run up with temperature the important variable. The machine was normally taken to speed in 15 minutes, run at desired speed for 15 minutes more, or until no further

temperature change was noted. Skin temperature was monitored by an in-house developed infra-red temperature sensor with a range 50F to 150F. When steady state temperature was realized, the particle generator was accuated and sampling continued for a period of 75 to 90 minutes when the generator was deactivated. The generator shut down was followed by 20-30 minutes of continued operation at speed thus pumping most of the particle bearing air from the system. This procedure was selected to avoid the problem of transient particulate deposit pattern evaluation. Establishment of the steady state pattern was considered necessary to any attempt at evaluation of pattern obtained when run up pumped aerosol rather than clean gas. The machine shut down was accomplished by sudden shut off of drive air, simultaneously shuting down the aerosol system air supply. Spin down from 12,000 rpm to 4000 rpm required only two minutes, the complete deceleration taking approximately six minutes. Dynamic operating characteristics, as represented by proximity probe data, were also monitored during all sampling runs. Photographic record of oscilloscope readout was made for at least the steady state speed and any unusual behavior was also similarly recorded. A dual trace Tektronix 564 memory scope was employed to permit holding the probe readout trace for amplitude measurement which could be compared to the values realized during earlier runs. The procedure outlined was followed for all sampling work conducted during the months of April, May and June.

A difficult task began upon removal of the sample collection surface. First operation, with simple geometry noted, was followed by immediate disassembly and removal of one-half the lining, the remaining half left installed. The machine was reassembled with plastic covers placed over the outlet and inlet openings to preserve the integrity of the remaining surface. A technique to strip the vellum into sections 2 cm wide, 75 cm long was used to reduce the large area to be handled. These strips were stored in a glass enclosed cabinet to prevent dust settlement from the work area environment. A strip was then further reduced to short segments which were mounted on microscope glass slides. Cover glass could not be used because of the risk of smearing the sample. Some thirty segments were studied under the microscope, the task a most frustrating one since slides lacked good optical flatness which caused focusing difficulty. Photos taken through the microscope were found to be useless because of this lack of optical flatness. Surface texture and features of the Vellum badly obscured particles, indeed made particle recognition virtually impossible. Vellum was abandoned in favor of a black Vinyl coated paper, handled in the same manner. While some particles were visible, the overall problems were much the same as noted for the vellum. An additional problem noted with simple paper or plastic coated paper, was a tendency to absorb moisture from the aerosol sampled. Since the Dautrebande generator produces water vapor with the solids entrained, the sample entering the centrifuge was very probably saturated with water vapor. Moisture absorption caused the paper to stretch under the influence of the centrifugal field and to pucker in places. The combined problems clearly eliminated paper from further consideration. A glass like surface was the obvious need, and thin glass was considered but rejected.

Mylar film was found to best approximate glass and film 0.005" thick was readily obtained with lithograph quality of surface. A full lining was again employed, the procedure used to produce microscope slides repeated. Optical flatness was still a serious problem, though much less so than noted with paper. Photo micrographs were again found too indistinct to be useful and careful visual study of many samples was required. With the Mylar surface employed, particles were discernable and it was possible to determine the presence of deposit almost everywhere within the centrifuge, with a region of relatively high density toward the lower inlet end. This experimental evidence was taken as indicative of a deep core penetration by the incoming aerosol and ruled out the possibility of a simple, open stub inlet. The open stub was then replaced with a stationary assembly which was basically made of two concentric cones of different radius. The outlet of this cone was an annulus 0.100 wide, with a mean radius equal 0.3 r_o, r_o the inner radius of the centrifuge bowl. Defining the ratio of inlet to bowl radius as α, consistant with the proposal upon which this work was based, this new geometry had $\alpha = 0.3$. A sampling run duplicating earlier open stub work with a speed 8000 rpm and subsequent study of collection surface indicated essentially no deposit above rotor mid plane. There appeared to be a tendency to band about one inch above the plane of inlet and there was clear evidence of back scatter into the region below the plane of inlet. This result indicated a need for rotation of the inlet geometry and with the stationary cone as a pattern, a rotating configuration was fabricaied. This change also required redesign of the inlet (lower) end cap. A photograph of this assembly is shown in Figure 18, with two pieces shown disassembled in Figure 19. The fully assembled centrifuge, as prepared for mounting in its support jacket is shown in Figure 20, the

Figure 18. Rotating Cone - Inlet End Cap Assembly

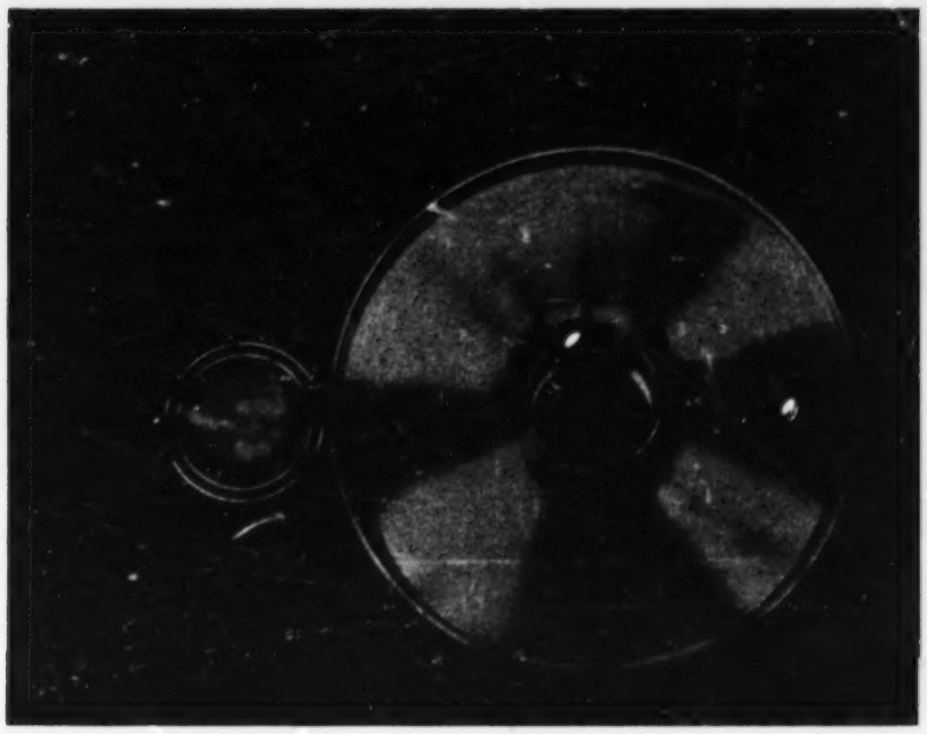


Figure 19. Disassembled Inlet End Cap - Rotating Cone

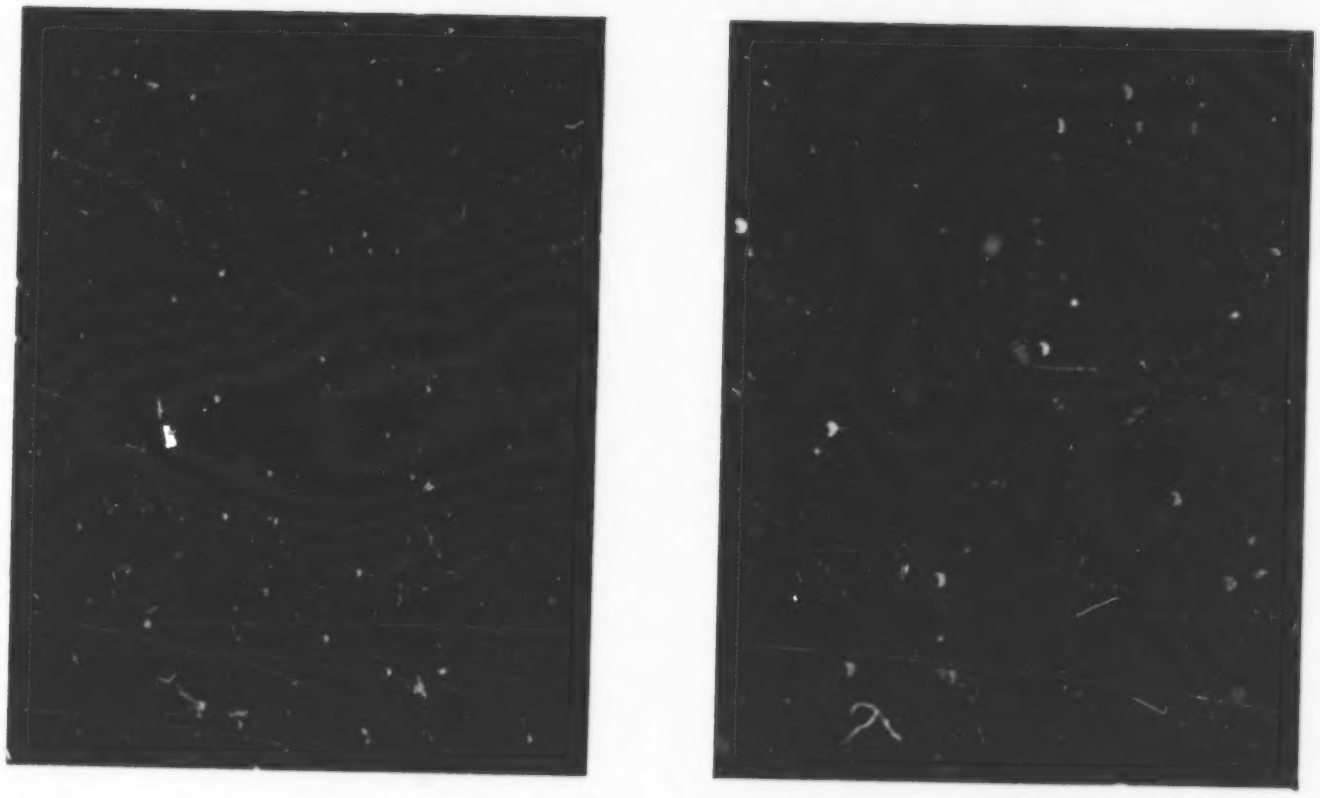
Figure 20. Fully Assembled Rotor

separated tube, and end caps in Figure 21. Experimental operation of this configuration indicated a sharp reduction of the area over which deposit was found, very little deposit noted above a plane 10 inches from the inlet. Severe evaluation limitations imposed by the lack of a deposit visible to the naked eye which required use of the microscope badly slowed the study. Rather obviously, a better technique was required.

The University's Department of Material Science had put its newly purchased scanning electron microscope in service during the first week of June. Dr. Kenneth Lawless, Professor of Materials Science kindly agreed to allow use of this new instrument for the purpose of scanning sample collection specimens. Samples were prepared using the appropriate specimen holders but the Mylar generated a static charge, when exposed to the electron beam, which totally obscured the field of view. Vacuum deposition of a conducting coating was attempted but proved to be unsatisfactory. Since the scanning microscope offered a means of relatively rapid surface study with excellent capability for photo readout, metallic collection surface was decided upon. Household aluminum foil, cut into strips 2.5 cms wide, was selected. Four such strips were equally spaced on the centrifuge inner surface and a sampling run carried out June 29, 1969. Strips were removed, cut into segments measuring 1 x 2 cm and mounted on microscope specimen holders with conducting high silver content cement. Controls available on the scanning microscope permitted traverse and rotation of the sample separately or simultaneously, thus providing for rapid overall scanning. The instrument has a resolution of 150 Angstroms and magnification capability to 60000x.

Figure 22a is a scanning microscope photograph, magnification approximately 1350x, taken of a small segment cut from one of the four aluminum foil sample collection strips. This specimen was cut 1/4" wide, 1/2" long, beginning at a

Figure 21. Rotor and End Caps



(a) 1350x

(b) 2700x

Figure 22. Scanning Microscope Photograph of Sample Collection Surface Located Approximately 3 3/4" Above Inlet End Cap. Particles 0.81μ

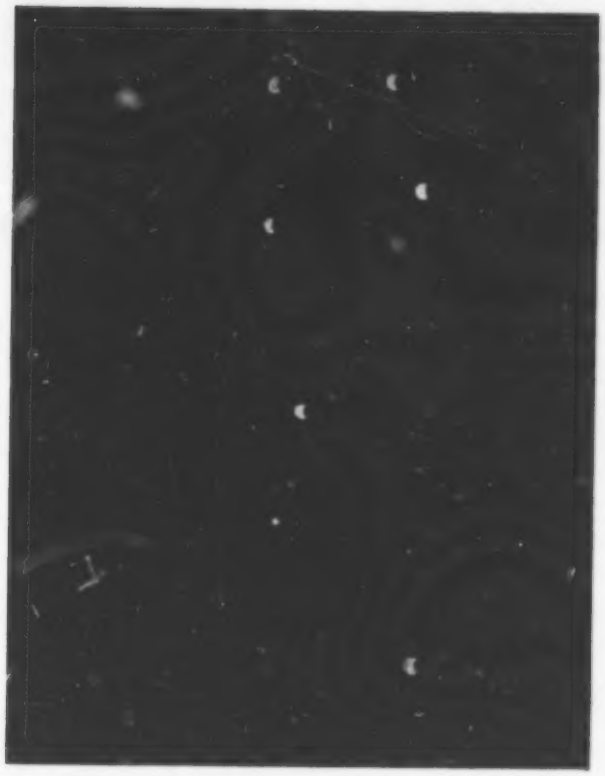


Figure 23. Scanning Microscope Photograph of Sample Collection Surface Located Approximately 4 1/2" Above Inlet End Cap. Particles 0.8lμ; 2700x

point 3 1/2" from the centrifuge end cap. The area, approximately 8×10^{-2} in.^{2} was estimated to be 3 3/4" from the end cap and 1 1/2" above the plane of gas entry into the centrifuge cavity. This specimen was selected as the region to examine based on predicted location of a band of 0.81μ particles for a centrifuge speed of 8000 rpm. Thirty-eight particles are visible including two doublets. A second photograph of this same specimen but with 2700x magnification is shown in Figure 22b. Fifteen particles can be seen including one doublet. The change of magnification required only turning a switch on the instrument control console. A similar photograph was made at lower magnification, in an effort to establish a statistically sound base for finding the spectral distributionband for this size particle. The difficulty can be readily emphasized upon examination of Figure 23, taken for a specimen immediately upstream of that shown in Figure 22 but at the same magnification. The area involved was approximately 3/8" from the first and only seven particles are visible. Actually no valid conclusion could be drawn from these specimens since the total number of particles visible was insufficient and this made evident a need for large numbers of particles in future experiments. No particularly useful information was gained relative to a detailed understanding of the internal flow pattern during this early flow experiment aside from verification of the serious effects to be expected from inlet geometry. Inlet geometry was expected to be a critical factor in this work and therefore, the results as reported here were considered encouraging. Subsequent work has served to further illustrate the importance of entrainment but this report has attempted to picture only the work performed to June 30, 1969.

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